

USING A "SOUNDTUBE" TO MEASURE NOISE OF STRUCTURAL SOURCES IN HIGH BACKGROUND NOISE ENVIRONMENTS

by

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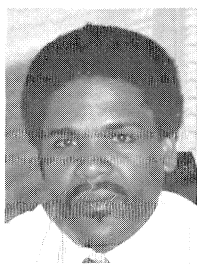
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ABSTRACT

Described in this paper is a device referred to as a soundtube, which has been designed to accurately measure only the normal component of the surface velocity of a vibrating structural noise source. Both laboratory and field data are presented. Significant advantages of the soundtube over an accelerometer for surface velocity measurements are:

1. Soundtube is not sensitive to high-level off-axis vibrations of the surface which do not contribute to sound radiation.
2. Since the soundtube is simply pressed against the surface, measurements can be made over surfaces which are lagged with insulation or are covered with ice.

The most useful application of the soundtube has been to estimate noise radiation from various structural noise sources

such as compressors and piping in environments with high background noise levels. By assuming that the source has a radiation efficiency which is equal to or less than unity, an upper limit of the noise radiation from each source component can be accurately determined.

INTRODUCTION

In a typical turbomachinery installation, the noise which exists at any particular location may be due to contributions from tens, if not hundreds, of individual noise sources. Although it is quite often possible to determine the major contributors of noise at various locations in the space by noting the character or quality of the sound, it is seldom possible to rank-order the various sources on a sound power basis using only microphone measurement results obtained around the machine perimeter. Most existing in-situ measurement standards call for microphone measurements to be performed at distances no closer than one meter from the source surfaces because of acoustic near-field effects. Perimeter microphone results which are obtained at these specified distances are very useful in determining the average sound pressure levels which exist in-situ. However, these data are due to contributions from all machine components, are affected by the acoustics of the environment, and are quite often "contaminated" by background noise sources such as piping.

In most, if not all, of the existing in-situ sound measurement standards, a background noise ambient test is required and is essential in order that the signal-to-noise ratio for the source under test may be determined. However, for most practical installations, the following limiting conditions are often found to exist:

1. The machine cannot be shut down for the ambient test, and, hence, there is no practical way of determining the signal-to-noise ratio or the contaminating influences of the background noise.
2. Even when the machine can be shut down for the ambient test, individual machine components and machine support equipment such as the gearbox, driver, etc., cannot be operated independently of the compressor for the ambient test. This is especially true and is a significant problem with attached piping.

Because of these practical limitations, microphone data taken at one meter can typically be used only to yield an upper-limited estimate of the machine noise levels when an ambient test cannot be run. Also, all noise sources which cease to

radiate when the machine is shut down must be considered to be an integral part of the source since they cannot be evaluated during the ambient test.

USE OF SURFACE VELOCITY TO ESTIMATE SOUND POWER

The sound power radiated from the surface of a structural noise source may be given in logarithmic terms as follows:

$$L_W = L_v + 10 \text{ LOG}(S) + 10 \text{ LOG}(\sigma_{\text{rad}}) \quad (1)$$

where L_W is the sound power level, L_v is the surface velocity level, S is the surface area of the radiator (in m^2/lm^2), and σ_{rad} is the radiation efficiency of the surface (see Appendix). The radiation efficiency can seldom be determined for sources in-situ, but will be equal to or less than unity for most practical, finite-size sources. Hence, the sound power level may accurately be determined from a slightly modified version of equation (1) as follows:

$$L_W \leq L_v + 10 \text{ LOG}(S) \quad (2)$$

Experience has shown that the estimated sound power level, based on equation (2), is very nearly equal to the actual sound power if: a) large sources are involved, b) the spectrum is controlled by predominantly high frequencies, and/or c) overall noise levels are A-weighted.

Accelerometer measurement techniques have long been used to determine the average normal component of the velocity over the surfaces of robust structural sources. Industry standards which are based on these accelerometer procedures for estimating the radiated sound power do not exist mainly because of the inequality sign which appears in equation (2) due to an unknown radiation efficiency. However, more serious practical drawbacks in the use of accelerometers include the following:

1. Many more accelerometer measurement positions are required than are necessary for perimeter microphone measurements.
2. In order to avoid system resonances in the audio range, accelerometers must be attached to the surface with an adhesive — a time-consuming process.
3. Accelerometer procedures are not feasible for use on surfaces which are thermally or acoustically lagged, or are covered with ice.
4. Extremely high vibration levels in a direction which is parallel to the surface may result in significant measurement errors due to cross-axis pickup in the accelerometer.

USING MICROPHONE SYSTEM TO MEASURE SURFACE VELOCITY

Shown in Figure 1 is a microphone inside a plexiglass tube which will be shown to yield accurate and repeatable estimates of the normal component of the velocity over the surfaces of robust sources. The device is pressed against the surface of interest and the circular area of the source under test attempts to radiate as if it were a piston located on an infinite flat plate. This is because the test area is not permitted to acoustically interact with the rest of the radiator. The acoustics of the cavity are not fully understood and it is not reasonable to assume that plane-wave propagation occurs over the entire audio frequency range. However, if the "soundtube" can be shown to yield repeatable results over a large variety of sources and is easily

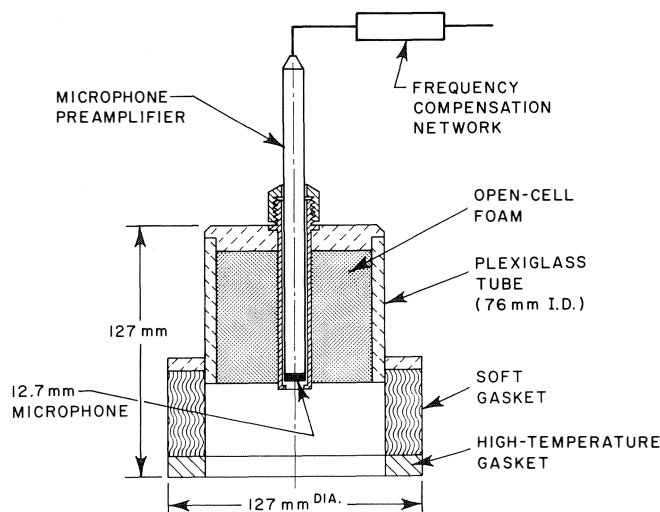


Figure 1. Shown are Views of the Enclosed Microphone System Known as the Soundtube. The Soft, High-Temperature Gasket Permits the System to be Used on Hot Surfaces with Diameters as Small as 20 cm (8 in) in Diameter.

calibratable as a surface velocity device, then it is not necessary to fully understand the acoustics of the cavity. The open-cell foam was added to the cavity because it was found from laboratory results to improve the high-frequency system response.

The soundtube has been evaluated both in the laboratory and in-situ on a number of structural sources. The soundtube frequency response is defined as the difference in the soundtube sound pressure levels and the surface velocity levels as

TABLE I. SIX STRUCTURAL NOISE SOURCE COMPONENTS USED TO EVALUATE FREQUENCY CHARACTERISTICS OF THE SOUNDTUBE. Sixty one-third octave frequency band spectra were obtained on two horizontally-split compressor casings and system components.

	No. of Positions	A-Weighted Levels	
		Surface (Soundtube)	@ 1m (Microphone)
Propylene Compressor Casing (25,000 kW, 3700 rpm)	15	92	106
Balance Piston Line (20 cm O.D.)	5	96	
Steam Turbine	4	90	
Butadiene Compressor Casing (1,100 kW, 6750 rpm)	20	73	91
Compressor Discharge Nozzle	10	77	
Compressor Support Beams	6	79	

measured with an accelerometer, $(L_p - L_v)$. The system frequency response has been determined from sixty one-third octave frequency band spectra obtained on the surfaces of the six structural noise source components listed in Table I. The computer-based measurement system shown in Figure 2 was used to obtain both accelerometer and soundtube data at each test location before proceeding to the next. In this way, the

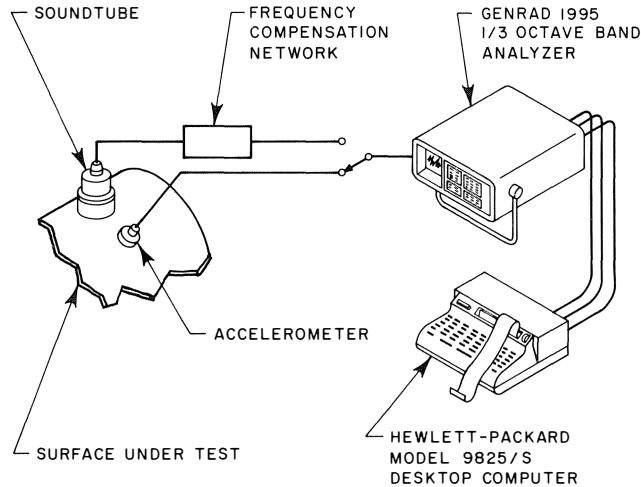


Figure 2. The Use of this Computer-Based, One-Third Octave Frequency Band, Real-Time Analysis System Permitted Laboratory-Grade Measurements to be Performed in situ.

computed differences in soundtube and accelerometer levels $(L_p - L_v)$ are not dependent on the long-term stability of the compressor system.

In order to determine the average frequency response and the statistical variation in the frequency response of the soundtube, mean-values and standard deviations of $(L_p - L_v)$ over the six sources were computed. All computations were performed on an octave band frequency basis, and the computed mean-values as a function of frequency are considered to be our most accurate estimate of the soundtube frequency response. The soundtube frequency response curve determined from the sixty spectra is plotted in Figure 3. The

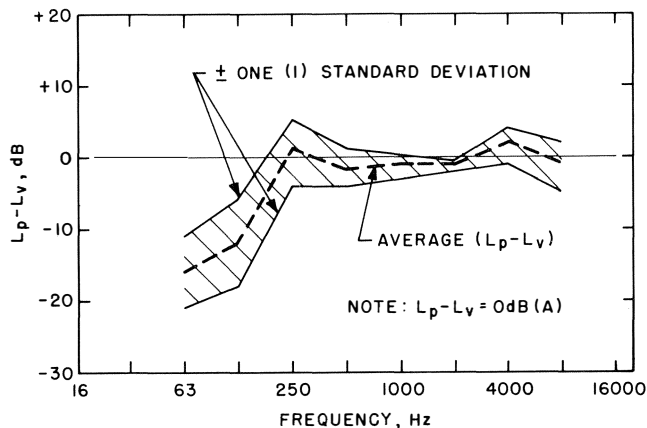


Figure 3. Soundtube Octave Band and A-Weighted Frequency Response as Determined from Data Obtained on Six Sources in situ.

average A-weighted $(L_p - L_v)$ response was also determined and was found to be equal to 0 dB (no correction required for overall A-weighted levels).

Although the computer-based measurement system shown in Figure 2 was used to calibrate the soundtube, the device was specifically designed for use with a portable sound level meter or octave band frequency analyzer using the microphone supplied with the instrument. This method has been used to obtain data on fifteen centrifugal compressors in the field in addition to the two units listed in Table I. Data from two full load factory tests of barrel compressors has also been obtained.

LABORATORY TESTS

Shown in Figure 4 is a large cylindrical noise source used to determine the soundtube frequency response over, and the radiation efficiency of, a large structural radiator under laboratory conditions. The cylindrical shell is 245 cm (100 inches) long, 91 cm (36 inches) in diameter, and has a thickness of 0.95 cm (3/8 inches). Chains attached to a rotating shaft inside of the cylinder were used to excite the shell, and the spectrum of the radiated noise was found to be very similar to typical spectra of noise radiated from centrifugal compressor systems in-situ.

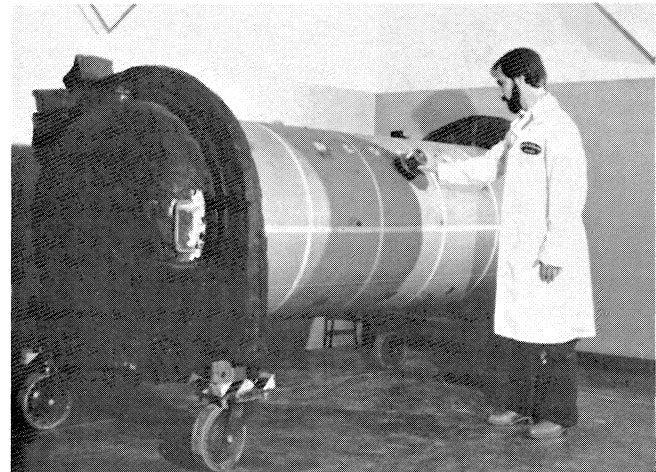


Figure 4. Demonstrated Use of the Soundtube Over the Surface of the Chain-Excited Cylindrical Shell.

A reverberation chamber which meets the specifications of ANSI S1.21 was used to determine the sound power radiated from the shell surface. All other parts of the structure were effectively lagged in such a way that the shell was the only significant noise source. The average acoustic surface intensity levels, L_I , were then determined from the computed sound power levels using the following equation:

$$L_I = L_W - 10 \text{ LOG}(S) \quad (3)$$

Shown in Figure 5 are plots of the average surface velocity levels over the shell as determined from twenty measurement locations and of the average surface intensity levels as determined from equation (3). The difference spectrum represents the radiation efficiency of the source, and even with the large differences which exist at low frequencies, the overall surface velocity level was found to be equal to the overall surface intensity level, on an A-weighted basis ($L_I = L_v = 103 \text{ dB(A)}$).

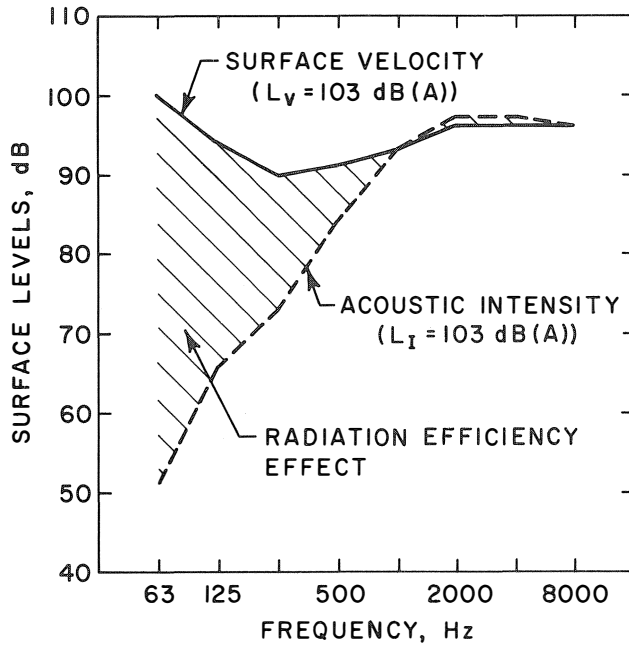


Figure 5. The Surface Velocity Levels were Obtained from Twenty Accelerometer Locations Over the Shell and Acoustic Intensity Levels were Obtained in the Reverberant Room. Differences Illustrate that the Shell has Unity Radiation Efficiency Above ~ 1000 Hz.

In Figure 6, both the average soundtube surface pressure levels and the average surface velocity levels are plotted relative to the average acoustic surface intensity levels. Whereas the surface velocity levels are shown to be significantly higher

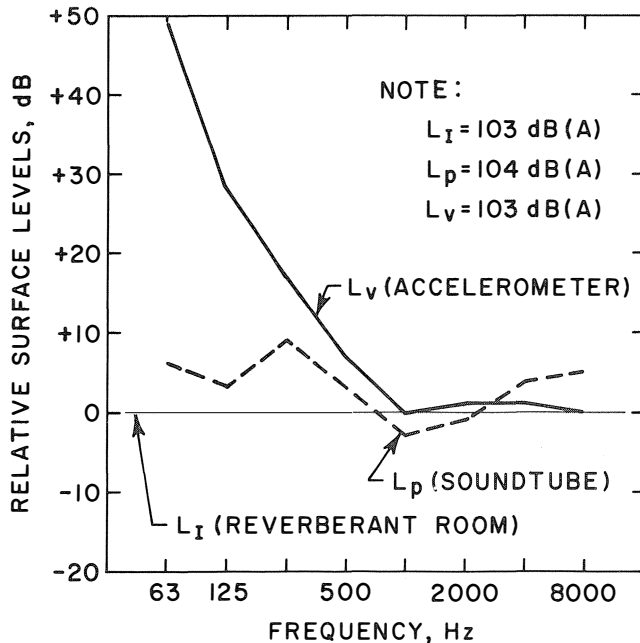


Figure 6. Shown are the Accelerometer (L_v) and Soundtube (L_p) Results Plotted Relative to the Acoustic Intensity Levels which were Determined from Data Obtained in the Reverberation Room.

than the surface intensity levels at low frequencies, the soundtube is shown to yield much better estimates of the surface intensity on an octave band basis. It is also important to take note of the fact that all three procedures are within 1 dB(A) on an overall basis.

CONCLUSIONS

In situations where existing industry standards cannot be used for machinery noise measurements in-situ — which will be the majority of cases for large centrifugal compressor systems — the soundtube may be used to provide an accurate and conservative estimate of the surface intensity levels of robust structural radiators. The average sound intensity levels, as seen in Table I, were found to be from 14 to 18 dB(A) below the average A-weighted sound levels obtained at 1 m distances around the machines perimeters. This again illustrates the degree of "contamination" which can exist near these large centrifugal compressors and demonstrates the ability of the soundtube to measure system components in high-background noise environments. The soundtube technique could also be applied to other turbomachinery and components.

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NOMENCLATURE

- L_W = sound power level, dB = $10 \text{ LOG}(W/W_{\text{ref}})$
 W = sound power, Watts
 W_{ref} = reference power, = 1. pW
 L_v = surface velocity level, dB = $20 \text{ LOG}(v/v_{\text{ref}})$
 v = surface velocity, m/s
 v_{ref} = reference velocity = 5×10^{-8} m/s
 L_p = sound pressure level, dB = $20 \text{ LOG}(p/p_{\text{ref}})$
 p = sound pressure, Pa
 p_{ref} = reference pressure = $20 \mu\text{Pa}$
 L_I = sound intensity level, dB = $10 \text{ LOG}(I/I_{\text{ref}})$
 I = sound intensity, Watts/m²
 I_{ref} = reference intensity = 1. pW/m²
 S = $\frac{\text{surface area of radiator, m}^2}{\text{Im}^2}$
 $\rho_o c_o$ = acoustic impedance of gas, rayls
 ρ_o = density of air, Kg/m³
 c_o = speed of sound in air, m/s
 $\langle \bar{X} \rangle$ = space/time average of variable X
 σ_{rad} = radiation efficiency of source, unitless

$$= \frac{W}{\rho_o c_o S \langle \bar{v}^2 \rangle}$$

APPENDIX

RADIATION EFFICIENCY OF STRUCTURAL RADIATORS

The sound power radiated by a structural noise source may be given by:

$$W = [\rho_0 c_0 \langle v^2 \rangle S] \sigma_{\text{rad}} \quad . \quad (\text{A-1})$$

The quantity in the brackets represents the sound power which would be radiated by a section of an infinite plate of area S and average velocity v . The radiation efficiency, σ_{rad} , is a multiplier which must be used to account for the reduction in radiated sound which will occur for any practical source of finite dimensions. In exceptional cases, a real source may radiate sound more efficiently than a piston in an infinite plate near the critical frequency of the surface of the source. Equation (1) may also be given in logarithmic form as:

$$L_W = L_v + 10 \text{ LOG}(S) + 10 \text{ LOG}(\sigma_{\text{rad}}) \quad . \quad (\text{A-2})$$

Rearranging equation A-2 and making use of the fact that the acoustic intensity level over the surface S may be given by:

$$L_I(\text{surface}) = L_W - 10 \text{ LOG}(S) \quad , \quad (\text{A-3})$$

the radiation efficiency may then be expressed as:

$$-10 \text{ LOG}(\sigma_{\text{rad}}) = L_v - L_I(\text{surface}) \quad . \quad (\text{A-4})$$

Therefore, since the radiation efficiency varies as $0 < \sigma_{\text{rad}} \leq 1$, the relationship of the velocity levels to the acoustic intensity levels over the source surface may be written:

$$L_v \geq L_I(\text{surface}) \quad (\text{A-5})$$

Equation A-5 reflects the fact that estimated surface acoustic intensity or sound power levels which are based on surface velocity measurements will usually be higher than the actual levels since the radiation efficiencies of most practical surfaces will be less than unity.

